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EXPERIMENTS in CONTROL of FLEXIBLE STRUCTURES WITH NONCOLOCATED SENSORS and ACTUATORS[1]

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Experimental apparatus has been developed for physically testing control systems for pointing flexible structures, such as limber spacecraft, for the case that control actuators cannot be colocated with the sensors. (An example is the Galileo spacecraft, whose television camera at one end of a flexible beam must be pointed by torquing at the other end of the beam). With colocation, good stable control is very easy to achieve. With noncolocation it is extremely difficult, particularly if structural damping is very low and spacecraft stiffness and inertia values are uncertain and changing, as they typically are. For the apparatus we have built, structural damping ratios are less than 0.003, each basic configuration of sensor/actuator noncolocation is available, and inertias can be halved or doubled abruptly during control maneuvers, thereby imposing, in particular, a sudden reversal in the plant's pole-zero sequence, a most difficult problem for the controller. First experimental results are presented, including stable control with both colocation and noncolocation. The inherent robustness of the former is clearly seen, as is the great difficulty of achieving robustness for the latter. (Schemes for doing so are now being explored, and future experiments will establish what the best achievable robust but nonadaptive control is, and will develop adaptive control .) What we hope to contribute here is a "red flag" warning about noncolocated control of flexible structures: there are configurations –indeed simple ones– for which there may be no practical alternative to adaptive control.

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1. INTRODUCTION

A crucial problem for some flexible spacecraft is that the location of points at their extremities must be controlled, sometimes to very high precision, by torquing at some other point (e.g., the spacecraft center) that is separated from the first by sections of flexible structure. We shall call this "noncolocated control" for short. (The sensors and actuators used for control are not colocated, but are separated by flexible structure.) This is an extremely difficult control problem: for the opportunities for instability in such closed-loop systems are many and fundamental.

A particular case in point – and the initial stimulation for our research– is the Galileo spacecraft, which will be sent in the mid 1980's to study Jupiter and its moons that Galileo discovered in the mid 1620's. The spacecraft spins slowly, but there is a beam-like structural system which is to be maintained inertially fixed by a motor at the hub. At the beam end is a television camera and other instruments which must be pointed, accurately and steadily on command, by a torque applied back at the hub and transmitted through the system of flexible beams to the television platform.

More generically, there will be large flexible spacecraft of many kinds on which the positions of many points at the spacecraft extremeties will need to be closely controlled, where the point positions can indeed be accurately measured (by optical means, for example), but where it is impossible or prohibitively costly to have an actuator at each point. Much greater design freedom is available if the technology is in hand to achieve precise, stable control by applying control torques at a distance, through the flexible structure.

This difficult control problem is further compounded by the facts that the physical damping in such large structures in space is apt to be very low, that the physical parameters are likely to be uncertain (prelaunch measurements at one g being difficult and inaccurate) and that the parameters will vary, sometimes by large amounts, as the spacecraft configuration is changed in the course of the mission.

It was discovered as early as 1965 (Refs 1,2,3) that in the simpler case of colocated sensors and actuators ("colocated control") one can guarantee stability with relatively simple control laws. Because of this property (presumably) nearly all of the theory to date for controlling flexible structures has begun by assuming colocation; and a considerable body of theory has indeed been developed for this case (Refs 3-8). Ref.4 does contain a design method that is also applicable for noncolocated sensors and actuators using output feedback; but the method does not address the real question of stability in the presence of parameter variations (for which, in fact, stability is *not* guaranteed). Ref.9 also presents a design for control of a flexible beam with noncolocated sensors and actuators. The effects of control and observation spillover were taken into account in the report; but, again, the serious question of performance when parameters vary was not considered. In short, the references cited above do not address the question of robust control of flexible systems using noncolocated sensors and actuators; and none presents experimental data. Reference 13 does deal directly with noncolocated control of a flexible beam, and presents experimental results. That work is complementary to the experiments with a multiple disk system presented here. Reference 10 addresses directly the Galileo problem of noncolocated control described above, and presents a form of adaptive control for solving it, including both theory and simulation.

The details of why colocation leads to simple control, and why noncolocation does not, will be described presently. The object of our research is to develop and demonstrate some of the control understanding required to solve this problem– to effect precise, stable control in the presence of large changes in parameters for the general case of *noncolocated* sensors and actuators – using extensions of control theory and laboratory experiments to do so.

For our first experiments we have concentrated on laboratory structures that would have very low inherent damping. This is of course particularly difficult to achieve with beams in a one-g field, and with air present; so we have developed the special apparatus we report on here, a torsional system with which we have been able to realize modal damping ratios of $\zeta \leq .003$. (We

are also continuing our studies and experiments on the control of flexible beams.) The torsional system leads one, in a most direct way, to several fundamental insights, as we shall discuss presently.

First we describe our experimental system.

2. EXPERIMENTAL APPARATUS

The laboratory system constructed for this investigation into control of noncolocated systems is shown pictorially in Fig.1 .

The system's "plant" consists of four steel disks nine inches in diameter with a thickness of one-half inch, attached firmly to a central, connecting torsion rod one eighth inch in diameter. The system is suspended from the ceiling via a long length of piano wire to thrust relieve the bearings . Each disk is instrumented with an angular position sensor. (One is shown.) A brushless DC torque motor is installed at the third disk station, and provision exists for a second motor to be mounted on the lowest disk. A digital minicomputer is used to implement the control algorithms developed for the four-disk system and to collect the experimental data. A general block diagram of the control system is shown in Fig.2 .

Since the central experimental focus is to demonstrate control of noncolocated systems with uncertain parameters, the top disk is made so that its inertia can be varied while the system is under closed-loop control. This was achieved as shown in Fig.3: The top disk consists of two pieces, an outer ring, and an inner disk, and a lifting mechanism can be used to separate the ring from the disk in a fraction of a second. (The slight conical taper on the disk periphery and ring inner surface ensure that very little force is needed to separate the two; yet the large contact area prevents slipping when the ring and disk are engaged.) The major, fundamental effect of this parameter change upon the open-loop transfer function from the actuator to the noncolocated sensor on the bottom disk will be examined in the next section.

The experimental design was governed by the requirement to incorporate : (1) sensor-actuator noncolocation, (2) a high degree of structural flexibility, (3) low inherent damping, and (4) the above-noted capability to make a large, sudden change in a key parameter. At the same time, we wished to construct a system whose dynamics are well understood, to insure unambiguous physical understanding of the closed-loop behavior.

Of course a large space structure differs significantly from our laboratory model. The presence of an infinite number of modes and high frequency modal uncertainty represent added complexity. However, the apparatus of Fig.1 permits early useful experimentation upon a simple physical system that retains the basic ingredients of sensor/actuator noncolocation and large parameter changes, and that achieves much lower damping than is possible with a laboratory beam.

Experimental results reported in this paper will show straightforward closed-loop control of the four-disk system in both a colocated configuration, Fig.4a, and a noncolocated arrangement Fig.4b, and the inherent inadequacies of the latter control will be demonstrated.

We begin by establishing the dynamics of the plant itself.

3. PLANT DYNAMICS

Fig.5a shows the plant (open-loop) experimental response to an initial condition containing primarily the first vibration mode and the "rigid body" mode. The damping ratio of the first vibration mode is seen from Fig.5a to be about $\varsigma_1 = 0.003$. (The frequency is seen to be 1.56 Hz.) A similar test, Fig.5b, shows the second vibration mode damping ratio to be about $\varsigma_2 = 0.002$. (The second mode frequency is seen to be about 2.9 Hz. The ten-second-period oscillation in Fig.5 occurs, rather than the infinite period associated with a true rigid body mode, because of the wire used to suspend the system.)

Fig.6a shows the transfer functions for the two colocated input-output arrangements (actuator

in to sensed displacement out) that are available on the four disk system. The pole-zero diagram for each system is displayed at the right side of the figure. A most significant feature of all such colocated systems is the pole-zero alternation in the transfer function, as one moves up the imaginary axis. This is the very desirable property that control designers have utilized so effectively for many years.

Fig.6b indicates the noncolocated transfer function from an actuator located at one end to a sensor at the opposite end. This arrangement leads to a transfer function with no zeroes.

Next, moving the actuator to a disk one removed from one end, but still sensing the position of the disk at the opposite end, Fig.6c, gives a transfer function with a single zero between the first and second vibration poles. A different value for a plant parameter, such as inertia or, stiffness could yield instead a plant transfer function with a zero between the second and third vibration poles. It is this fundamental reversal in pole zero sequence (which we call "zero flipping") that makes stable control so particularly difficult to achieve in noncolocated control.

System identification tests were performed upon the four disk system to confirm the capability for the system to demonstrate a pole-zero flip. Two different techniques were found useful for this: sine wave tests and white noise tests using adaptive lattice filters. The sine wave tests are useful for directly identifying poles and zeroes of the four disk system. This test was performed using a sine wave generator, frequency counter, and laboratory oscilloscope. For these tests, the sensor-actuator configuration of Fig.4b was used. The system was driven at a frequency near a structural mode and a Lissajou figure displayed on the oscilloscope, using the sensor output on one channel and the plant input signal on the second channel. The vibration mode frequency could be determined very accurately in this way. This is true because the vibration modes are very lightly damped, so that at a resonance the Lissajou ellipse will have a 90 degree phase shift between input and output, because the plant ouput will be maximized. The driving frequency could then be 'dialed in' to find the vibration mode

frequencies. In practice, it is very easy to align the ellipse by varying the frequency until a stationary, vertical image is obtained. The transfer function zeroes can be found in this way by varying the driving frequency to null the plant output.

The second test procedure involved the use of a laboratory test instrument, a Genrad 2515 Structural Analyzer. This device is actually a minicomputer equipped with real-time data acquisition hardware and software. This kind of instrument provides somewhat of a 'black box' approach to system identification, and will likely come into more and more prominence for structural testing via computer aided methods, due to their high utility. This test does not measure transfer function poles and zeroes directly. Rather, a data batch is gathered and then processed via a filtering algorithm to fit a linear transfer function to the data.

Test Results

Using the previous two test techniques the following poles and zeroes were identified:

For $J_4 = 1.0$ the sine wave test indicated $\omega_1 = 9.86$ (rad/sec), $\omega_2 = 18.22$, and $\omega_3 = 23.81$. The zero was measured to be $z_1 = 12.89$. The Genrad equipment gave the following: $\omega_1 = 9.9$, $\omega_2 = 18.0$, and $\omega_3 = 24.0$. The zero was found to be $z_1 = 13.0$ The two tests differ by less than 1.0 percent. The ratios of these values match precisely the theoretical ratios of Fig.6c.

For $J_4 = 0.25$ the sine wave test indicated $\omega_1 = 11.94$, $\omega_2 = 21.48$, and $\omega_3 = 29.34$. The zero was measured to be $z_1 = 25.9$. The Genrad test measured $\omega_1 = 12.0$, $\omega_2 = 22.0$, and $\omega_3 = 30.0$. The zero was found to be $z_1 = 24.0$ In this case the difference between the tests is less than two per cent for the vibration frequencies, and four per cent for the zero frequency. Both tests confirm the pole-zero flip.

4. Control Design

To provide a clear context for the experimental results of part 5, we discuss next control design techniques for the four-disk system (using the root-locus format for exposition).Our

approach has, of course, much in common with that required for controlling flexible beams.We first discuss control for the easier case that sensor and actuator are colocated.

a. Colocated system

The transfer function from the colocated actuator and sensor on the third disk is

$$\frac{y(s)}{u(s)} = 152.\frac{(s^2 \pm 20.85j)(s^2 \pm 12.886j)(s^2 \pm 7.96j)}{(s^2)(s^2 \pm 23.81j)(s^2 \pm 18.22j)(s^2 \pm 9.863)} \tag{1}$$

A simple lead network can readily be made to stabilize this system, as shown in the root-locus plot of Fig.7 . The lead network chosen has the transfer function

$$\frac{u(s)}{y(s)} = -8.3\frac{(s+6.50)}{(s+33.0)} \tag{2}$$

The root-locus plot is drawn versus overall loop gain. The closed-loop system roots (indicated on the root locus) show that the rigid body mode is damped by fourteen percent, the first vibration mode is damped by twelve percent, the second mode is damped by two percent, and the third mode is damped by twenty-six percent. The closed-loop bandwidth is one Hz. (6 rad/sec), or two thirds of the first vibration mode: a reasonably fast system.

What is even more important is the inherent robustness of this control system. This is suggested from the root-locus, for the root-locus lies entirely in the left-half, or stable region of the s-plane. The lead network is effective for providing fast, robust control for the colocated system because the colocated transfer function (from torque to motion at the sensor) always has alternating poles and zeroes (Fig.6). *Thus, it is easy to phase stabilize every vibration mode without accurate knowledge of the vibration-mode frequencies or damping ratios.*

This result is clear on physical grounds: we know that the plant can be stabilized by adding passive dashpots . A dashpot removes energy from the system by providing a force proportional to rate, and the force is applied at the same point where the rate occurs. Using pure rate feedback would perform exactly the same function, cancelling one of the rigid body poles and

yielding a root locus in which the departure angle at every pole would be 180 degrees. The actual lead network is part of an active controller that causes the closed-loop system to behave nearly like a passive system, providing colocated rate feedback plus position feedback to control the rigid body mode. (The pole in (2) makes the control realizable by providing high-frequency roll-off.)

The simple lead network thus yields stable compensation for large parameter variations, and with a relatively high closed-loop bandwidth.

b.noncolocated system

Turning now to the noncolocated case, we find a much different situation. To understand it, we will compare the results of employing two kinds of compensator: first a lead network (as for the colocated case above), and then a full order optimal compensator.

The nominal plant transfer function is

$$\frac{y(s)}{u(s)} = 827,000. \frac{(s^2 \pm 12.886j)}{(s^2)(s^2 \pm 23.81j)(s^2 \pm 18.22j)(s^2 \pm 9.863)} \tag{3}$$

The salient new feature, with which the controller must now cope, is the striking effect that a parameter variation can produce in the plant transfer function of a noncolocated system. The effect is illustrated in Fig.8: as the value of inertia J_4 is decreased, a pole zero 'flip' can occur! That is, the transfer function zero that begins between the first and second vibration poles in the system with J_4 nominally unity moves to a new location between the second and third vibration poles when J_4 is reduced to 0.38 per cent of its nominal value or less. Since in general the nominal value of J_4 could actually be near 0.38 to start with, a very small pecentage change can in fact cause a pole zero flip. (For the experimental results reported here, J_4 was varied between its nominal value, one half its nominal value, and one fourth its nominal value. As discussed in a previous section, a pole-zero flip occurs in the four disk system when J_4 is varied between one half and one fourth of the nominal value.) This poses a most difficult control problem,

implying the need for a compensator that can provide over 180 degrees of phase margin near the frequency range where the pole-zero flip occurs. Thus, a controller designed for one value of J_4 will surely go unstable if J_4 subsequently takes on the other value, unless the possibility of the pole-zero flip is accounted for meticulously in the control design. And that is not easy.

"Pole-zero flipping" can be serious. It is true, of course, that if a small parameter change can cause a pole-zero pair to flip, this implies the pole and zero were close, or nearly cancelling in the open-loop system from control torque to sensor motion, so that this mode therefore does not contribute heavily to the system response to a command. But commands are not the only system input! There will also be disturbances acting on the structure. Moreover, because inherent damping is so low in large space structures, the pole need not be shifted very far to become unstable. To reduce the plant model by truncating this mode may well be to ask for disaster: for in the presence of even mild parameter uncertainty, the unstable mode may be discovered for the first time after the spacecraft has been launched. Instead, the control system should be designed to have a stabilizing effect -or at most no effect- on every pole, even if it is only possible to improve the pole location a small amount because of the nearby zero.

Lead compensation

Using lead compensation, we can in fact design a very-low-bandwidth controller for the noncolocated system. The control design basically treats the plant like a rigid body. The crossover frequency is chosen so that even the first vibration mode is gain stabilized. To achieve a stable system we must rely entirely on inherent damping in this case, because the lead network always destabilizes at least one vibration mode, Fig.9. Further, we must have a good idea of the vibration mode frequencies and damping ratios even when designing a low bandwidth controller.

The control design is essentially a trade-off to see how far the rigid-body roots can be moved before the flexible modes are destabilized: and the result will inevitably be slow control. This situation is thus markedly different from the colocated case, even for a low performance design.

Specifically, the lead network used in Fig.9 has the transfer function

$$\frac{u(s)}{y(s)} = -.389\frac{(s+0.2024)}{(s+.6692)} \tag{4}$$

In the resulting closed-loop system, the modified rigid body poles are at $s = -0.29, \pm 1.75j$, only about 17% of the first plant mode frequency, compared with 66% for the colocated case (Fig.9).

To achieve any higher performance in the noncolocated case, a higher order compensator must be used. Then faster response can be achieved; but robustness will be quite unacceptable, as we shall see.

LQG design

An optimal control design, using LQG synthesis techniques can yield a higher performance system, *if the plant parameters are precisely known*. The compensator will be $8th$ order, the same order as the plant to be controlled. The root locus of Fig.10 shows that the compensation consists of lead at low frequency and notch filters at each structural mode. (Ref. 12 provides a complete discussion about compensator transfer functions in optimally-controlled systems.)

It is the use of structural notches that allows the closed-loop system to achieve higher bandwidth than the simple lead network. Each notch consists of a pole-zero pair. The compensation basically cancels the structural pole with the notch zero and substitutes a more heavily damped compensator pole in its place. This allows the compensator to have higher gain and thus move the rigid body poles further. On the root-locus plot the effect of the notch filters is characterized by the fact that each plant structural pole moves to the left, or towards the stable region of the s-plane, while the notch poles move to the right. Again, the root locus was drawn by varying the compensator gain. The compensator transfer function in the example is

$$\frac{u(s)}{y(s)} = -124\frac{(s+4.88\pm 21.9j)(s-1.613\pm 20.6j)(s-.714\pm 10.4j)(s+1.63)}{(s+12.259\pm 7.043j)(s+2.78\pm 24.454j)(s+4.22\pm 19.17j)(s+1.069\pm 12.176j)} \tag{5}$$

The closed-loop poles are: $s = -2.0 \pm 23.9j, -2.2 \pm 18.5j, -1.5 \pm 9.8j, -4.5 \pm 2.8j$. The closed-loop bandwidth is 45% of the first vibration mode.

However, to be effective the notches must be tuned very precisely. The resulting controller is therefore sensitive to any change in a plant parameter, that is, to a change in either the plant's vibration frequency or its damping. That is, when the model is correct, the control system should meet it's specs; *but only so long as the system's parameters stay close to their assumed values.*

If they do not, the closed-loop system may well become unstable. Fig.11 is a plot of the locus of closed-loop roots when the compensator (Eq. 5) designed for the nominal system ($J_4 = 1$) is applied to a system in which $J_4 = 0.25$. In this case, the pole-zero flip causes the closed-loop system to become unstable well before the nominal compensator gain is reached. In fact, the first vibration mode is destablized immediately.

The nominal compensator can in fact stabilize the system only *when J_4 is within ten per cent of the design value,* and even then closed-loop performance is severely lowered.

Good noncolocated control can be very hard to achieve.

5. Experimental Results

We now present results from some physical experiments which demonstrate, some perhaps for the first time, each of the above fundamental concepts about flexible spacecraft control, namely: (1) ease of achieving fast, robust control with colocation, (2) the great difficulty –with noncolocation– of achieving control that is even stable: how slow such control must be, and how small a change in plant parameters can make it unstable.

Figs.12-14 show the behavior of the laboratory four-disk sytem under the several forms of closed-loop control described in Section 4.

Colocated Control

Nominal Plant

Fig.12a shows the response to an initial condition of the system in which closed loop control is achieved using colocated sensor and actuator. The system was initially excited by simply rotating it off zero and shaking one of the disks by hand to excite the rigid body mode and the first vibration mode. The system is uncontrolled during the first three seconds of the figure, at which point the control is suddenly turned on. The commanded position is zero, so the figure shows the regulation capability of the controller. The system natural frequency is 1.2 Hz., or about seventy percent of the first vibration-mode. This agrees quite well with the design given by the root locus of Fig.7. The response dies out in just over three vibration cycles. There is a small amount of second and third mode contained in the output, as well as some evidence of output quantization.

Fig.12b shows the response to an initial condition of the system which contains primarily the second vibration mode.

Step Response

Fig.12c shows the response of the same system to a ten degree step change in commanded position.The system natural frequency can again be seen to be about 70 percent of the first vibration mode, which is in agreement with the design given by the root locus of Fig.7 . The damping ratio is approximately twelve percent, also close to the predicted value of fourteen percent.

Affect of Parameter Change

Fig.12d shows the response of the four-disk system in which J_4 is only one fourth the nominal value, but using the lead compensation designed for the nominal system. The system response is essentially identical to the nominal case, thus demonstrating the robustness and high performance obtainable (Fig.8) when the sensor and actuator of the control system are

colocated.

Noncolocated Control

Figures 13 and 14 show the response of the four disk system in which closed loop control is achieved using *non*colocated sensor and actuator .

Simple Lead Controller: Nominal Plant

Response with the simple lead compensation of Fig.9 is shown in Fig.13a . In this case a bandwidth of only about ten percent of the first vibration frequency is possible, even when the plant parameters have exactly their nominal values. This agrees with the prediction of Fig.9. The response contains a component at the first vibration mode which does not die out perceptibly in ten seconds. This indicates the predicted inability of the low bandwidth controller to damp the vibration modes. (See Fig.5, the plant's free response).

Affect of Parameter Change

Fig.13b shows the effect of a parameter change upon this system's stability. In this case the system barely remains stable when J_4 is changed from its nominal to one half its nominal value, even for the extremely slow system achieved in Fig.13a.

"Optimal Controller"

Fig.14 shows experimental performance when the eighth-order LQG compensator of Fig.10 is used. Fig.14a shows the closed-loop step response. The response has about ten percent overshoot and a rise time of about one second. The steady-state performance has somewhat more "jitter" than did the colocated lead network; but it is much faster than the response obtainable with the noncolocated lead network. (The jitter could be reduced at the expense of slower response by adjusting the weighting factors.)

Affect of Parameter Change

The performance shown in Fig.14a is available from the LQG compensator *only* when the

plant parameters are precisely known. This is demonstrated dramatically in Fig.14b, which shows the unstable response when the LQG compensator designed for the system with $J_4 = 1.0$ is applied to the system in which $J_4 = 0.25$. The controller was turned on with the system in its nominal configuration. After two seconds the parameter J_4 was changed while the controller was on, and the subsequent rapidly growing unstable motion recorded. The frequency of the unstable vibration is 1.78 hz. The predicted value (from the root locus of Fig.10) is 1.8 hz. The initial rate of growth is also very close to the predicted rate: 0.9 seconds observed versus .93 second doubling time predicted. The LQG system is not at all robust in this case, as is shown dramatically in Fig.14b where, clearly, the closed loop system becomes unstable when the value of J_4 is changed.

In a subsequent series of experiments we will focus directly on the stability vs. robustness question. We will seek to establish the absolute best – i.e. most robust – nonadaptive control that is achievable, especially in the case where a pole-zero flip can occur. (These experiments will employ some new approaches to fixed compensator design.) With this result as a base, we will then begin to apply adaptive control techniques to achieve, finally, performance that is acceptable, even with a pole-zero flip.

6. Summary and Conclusions

This paper has described a new experimental apparatus for investigating control laws for large flexible spacecraft. The initial series of experiments have been intended to demonstrate the difficulties associated with active control of large space structures, particularly when the sensors and actuators are noncolocated. Such systems will have many low-frequency vibration modes, and very low inherent damping. The control system will be designed using a model of the structure which contains uncertainty, and the actual plant parameters will vary with time, so that the control system needs to be robust.

The laboratory system was designed to provide a control-system test bed exhibiting each of the above characteristics. The system possesses three vibration modes plus a rigid body mode,and is instrumented to allow control configurations with either colocated or noncolocated sensors and actuators. A key system parameter can be changed while the system is under closed loop control, so that robustness of the control design can be most severely tested. Natural damping of the system's vibration modes is less than 0.3%

What we have shown in the initial experiments reported here is that in the case where sensors and actuators are noncolocated, any controlled flexible system may well be extremely sensitive to the actual values of system parameters, so that quite sophisticated techniques are going to be needed to achieve fast, stable, robust control. When the sensors and actuators are noncolocated, the control system needs to account for the presence of many vibration modes. Modal damping ratios and vibration frequencies need to be known accurately or continually identified, because the controller will invariably destabilize some of the high frequency modes even when the plant is known, so that the typically low values of inherent damping will greatly limit achievable performance.

Finally, systems with sensor-actuator noncolocation can exhibit "pole-zero flipping" when parameters vary (while colocated systems always have alternating poles and zeroes,even when parameters vary greatly). It is suggested that control system designers be most wary of these conditions.

The next series of experiments will apply parameter optimization tools to investigate the capability of the most advanced robust-control design techniques to cope with such difficult problems as large parameter changes and pole zero flips. This work will provide a baseline to assess definitively the circumstances in which only adaptive control techniques can supply robust control for flexible spacecraft. One possibility for such adaptive control is suggested in Ref.10, and Ref.14 reports demonstrations of such adaptive control schemes applied to a simple

torsional system. The laboratory four-disk system described in this paper will also be modified to create a system with nearly equal vibration frequencies. This case presents an extreme challenge for adaptive control methods which rely upon frequency identification methods, and is thus very relevant for future work in control of large space structures.

REFERENCES

1. Gevarter , W.B., "Attitude Control of a Flexible, Spinning, Toroidal, Manned Space Station," Ph.D. Thesis, Dept. Aeronautics and Astronautics, Stanford University, Stanford, Ca., 1965.
2. Gevarter, W.B., "Basic Relations for Control of Flexible Vehicles," AIAA Journal, Vol. 8, No. 4, Apr. 1970
3. Martin, G.D., "On the Control of Flexible Mechanical Systems," Ph.D. Thesis, Dept. Aeronautics and Astronautics, Stanford, Ca., SUDAAR 511, May 1978
4. Lin, J.G., Hegg, Y.H., and Keat, J.E., "Output Feedback Control of Large Space Structures: An Investigation of Four Design Methods," Dynamics and Control of Large Flexible Structures, Proceedings of the First VPI &SU/AIAA Symposium, 1977
5. Elliott, L.E., Mingori, D.L., and Iwens, R.P., "Performance of Robust Output Feedback Controller for Flexible Spacecraft," Proceedings of the First VPI &SU/AIAA Symposium, 1977
6. Kammer, D.C., Sesak, J.R., "Actuator Number versus Parameter Sensitivity," Proceedings of the First VPI &SU/AIAA Symposium, 1977
7. Sevaston, G., Longman, R.W., "Optimal Positive Real Controllers," Proceedings of the Third VPI &SU/AIAA Symposium, 1979
8. Sesak, J.R., Likins, P.W., Cordadetti, "Flexible Spacecraft Control by Model Error Sensitivity Suppression," J. Astronautical Sciences, Vol 27, No.2, 1979.
9. Balas, M.J., "Direct Output Feedback Control of Large Space Structures," J. Astronautical Sciences, Vol 27, No.2, 1979
10. Kopf, E.H., T.K. Brown, and E.L. Marsh, "Flexible Stator Control on the Galileo Spacecraft," AAS Conf., paper No. 79-161, Provincetown, Mass., Jun. 1979.
11. Bryson, A.E., "Some Connections between Modern and Classical Control Concepts," J. Dynamic Systems, Measurement and Control, Vol. 101, 1979.
12. Wie, B., "On the Modeling and Control of Flexible Space Structures," Ph.D. Thesis, Dept. Aeronautics and Astronautics, Stanford, Ca., SUDAAR 525, June 1981
13. Cannon, R.H., Jr.,and Schmitz, E., " End point Control of One-link Flexible Manipulators", To be published in Int. J. of Robotics, 1984
14. Rosenthal, D.E., and Cannon, R.H., Jr., "Experimental Results in Adaptive Control of Flexible Structures With Noncolocated Control", To be published in J. of the Astronautical Sciences, 1984

LIST OF FIGURE CAPTIONS

FIG.14b OFF-NOMINAL PLANT WITH NOMINAL COMPENSATOR,

SHOWING INSTABILITY

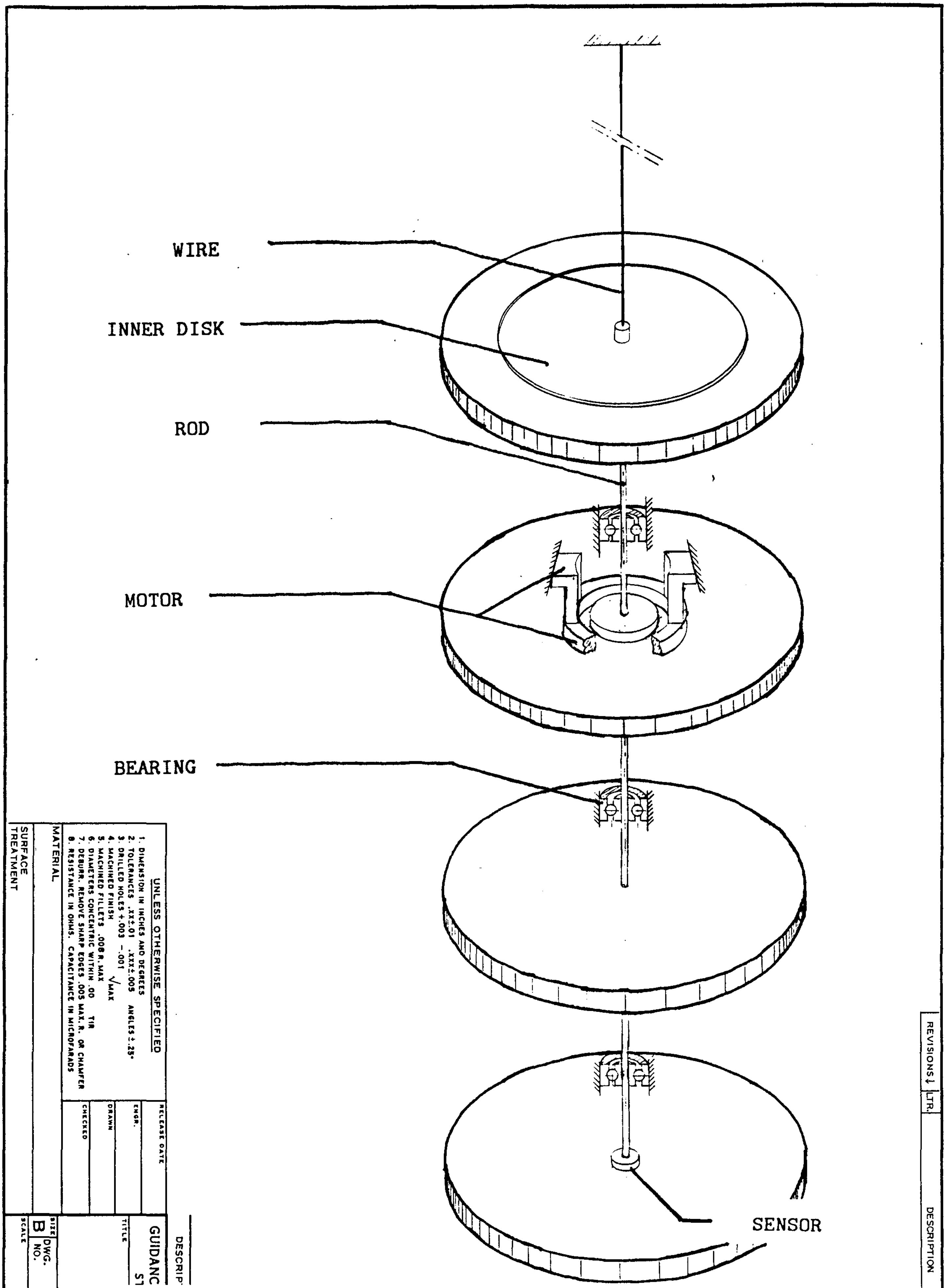


FIG.1 LABORATORY FOUR-DISK SYSTEM

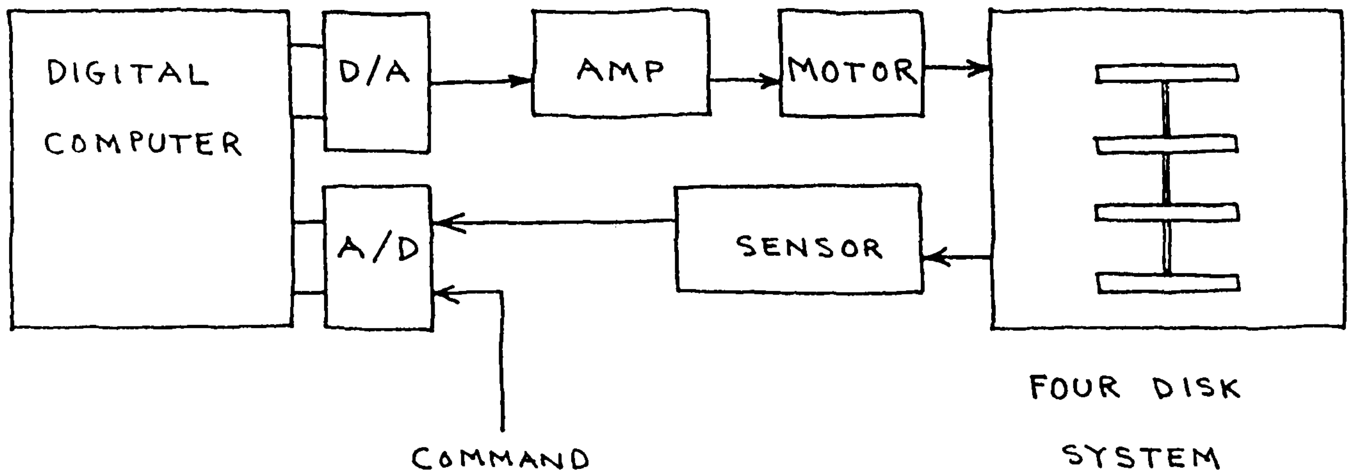


FIG.2 LABORATORY SYSTEM BLOCK DIAGRAM

REVISIONS↓	LTR.	DESCRIPTION	DATE	APPROVED

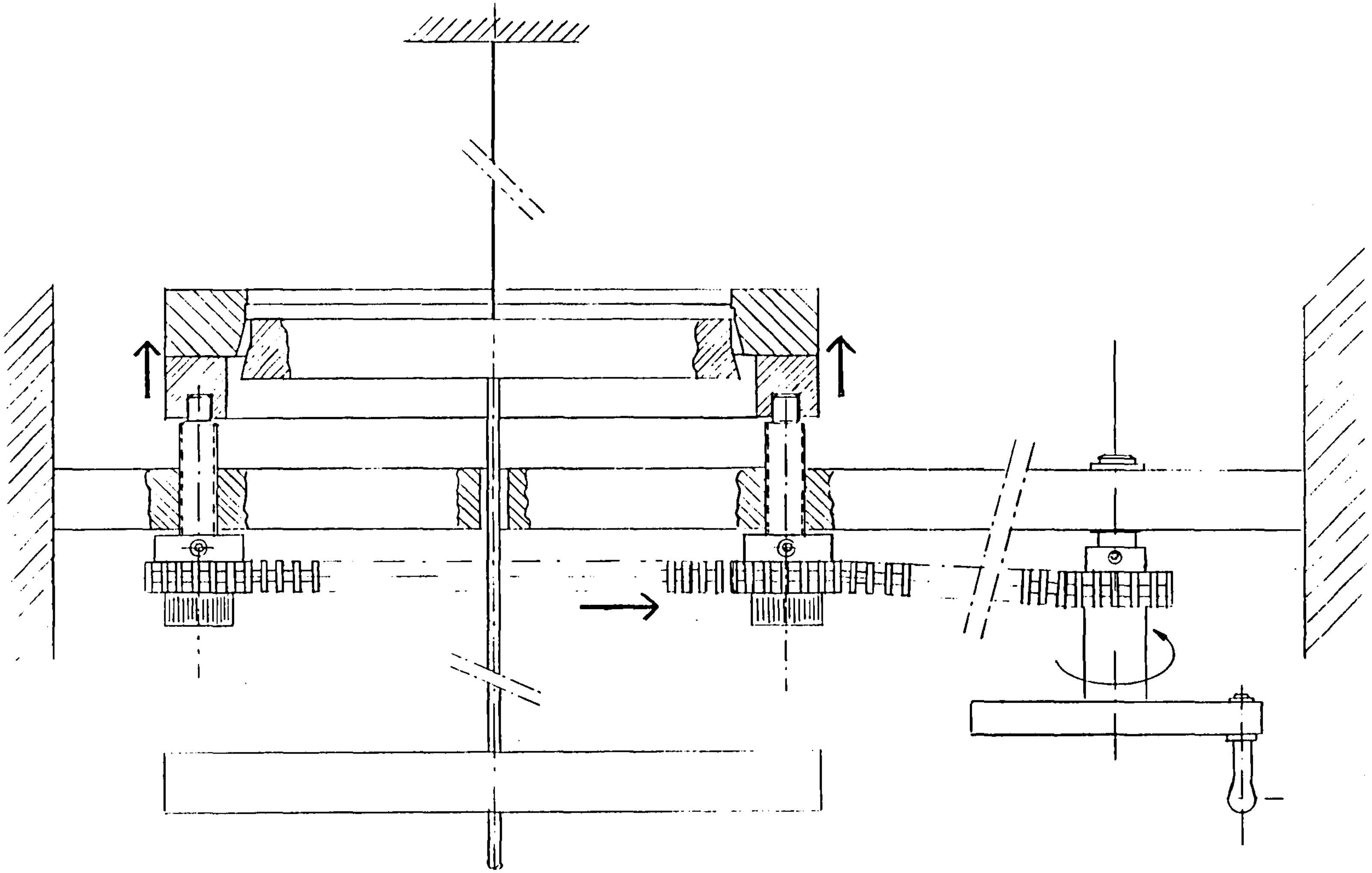


FIG.3 MECHANISM TO CHANGE DISK INERTIA ABRUPTLY

DESCRIPTION	DWG. OR PART NO.	NO. REQ'D	ITEM NO.

UNLESS OTHERWISE SPECIFIED

1. DIMENSION IN INCHES AND DEGREES
2. TOLERANCES .XX±.01 .XXX±.005 ANGLES ±.25°
3. DRILLED HOLES +.003 −.001
4. MACHINED FINISH √MAX
5. MACHINED FILLETS .008 R. MAX
6. DIAMETERS CONCENTRIC WITHIN .00 TIR
7. DEBURR, REMOVE SHARP EDGES .005 MAX. R. OR CHAMFER
8. RESISTANCE IN OHMS. CAPACITANCE IN MICROFARADS

RELEASE DATE

ENGR.

DRAWN

CHECKED

MATERIAL

SURFACE TREATMENT

GUIDANCE & CONTROL LABORATORY
STANFORD UNIVERSITY

TITLE

SIZE B DWG. NO.

SCALE

WEIGHT

SHEET OF

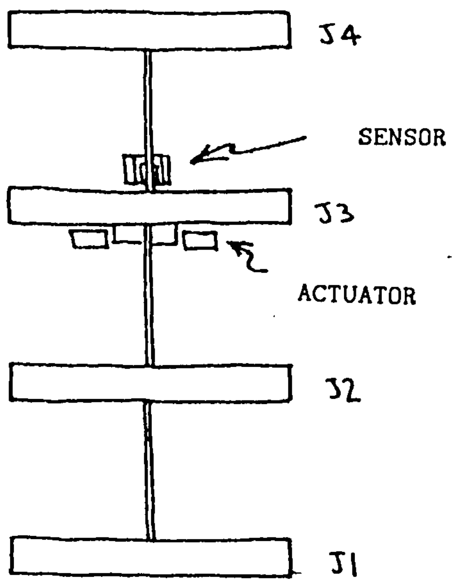


FIG.4a COLOCATED

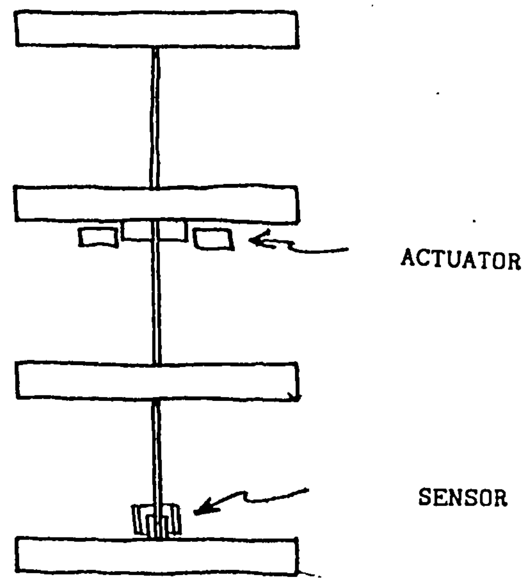
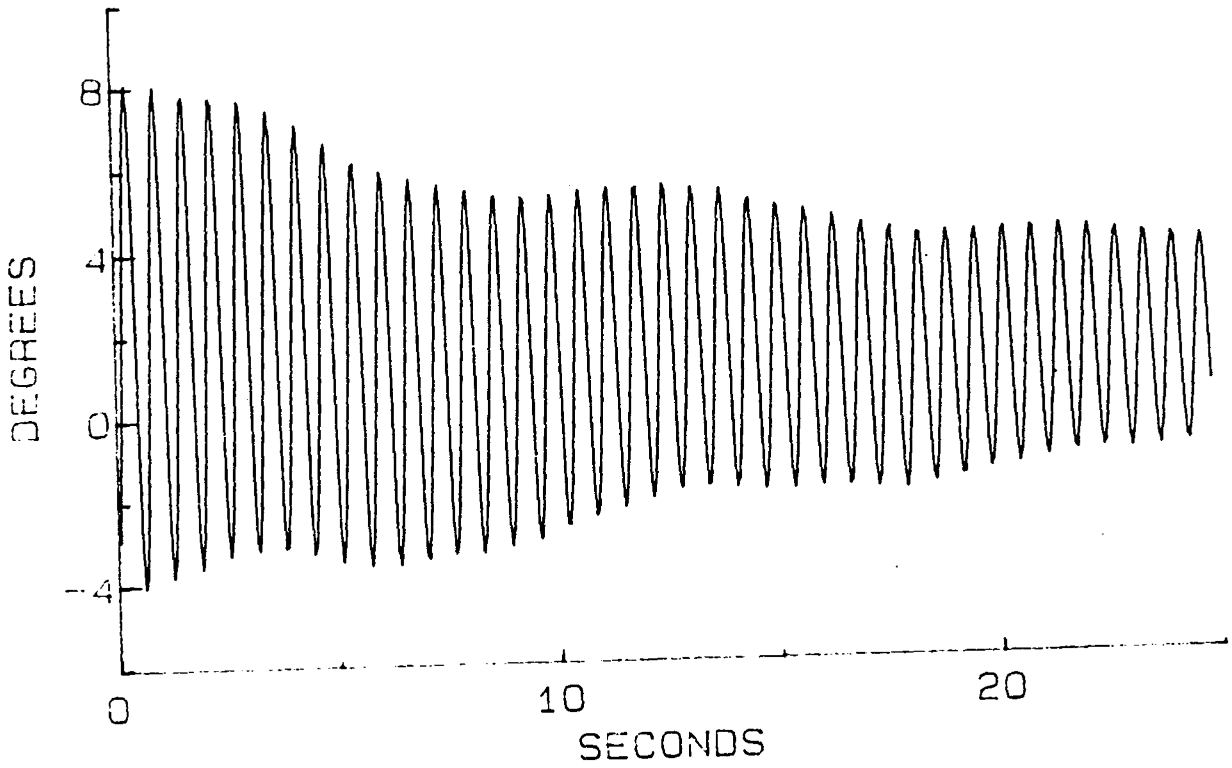


FIG.4b NONCOLOCATED

FIG.4 SENSOR ACTUATOR ARRANGEMENTS



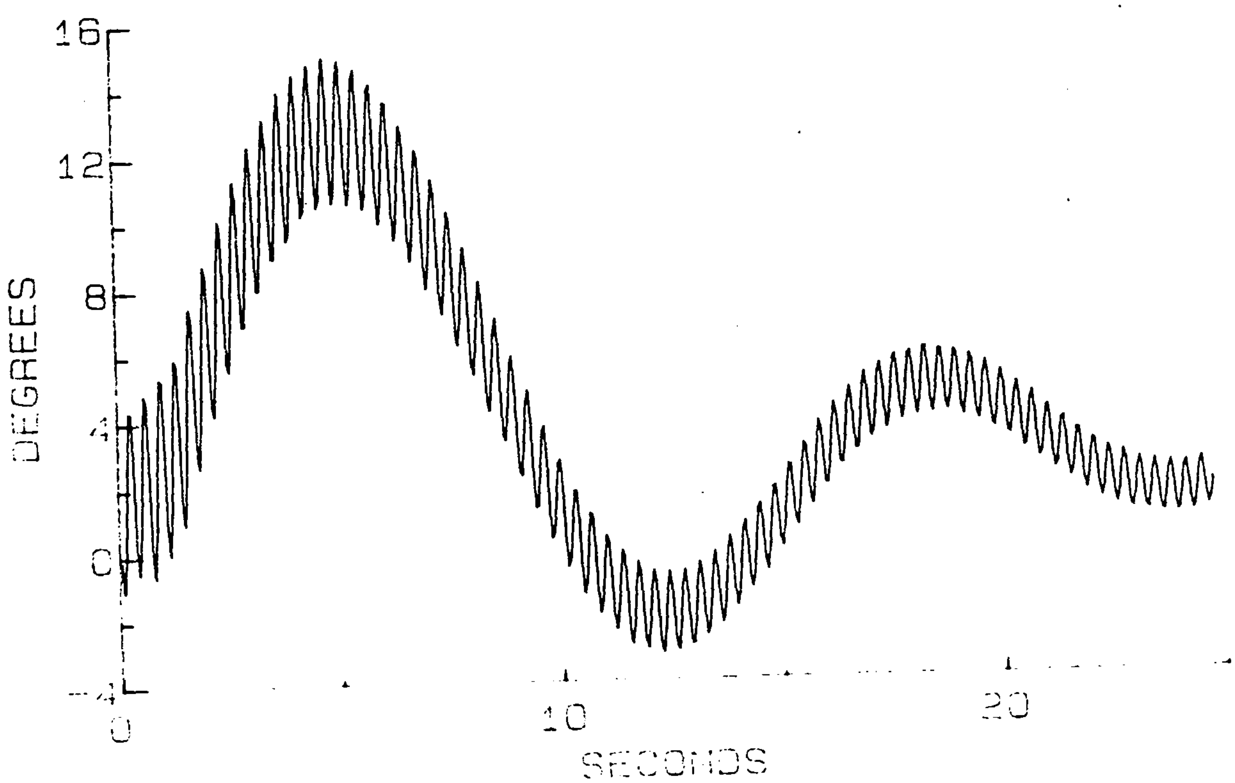
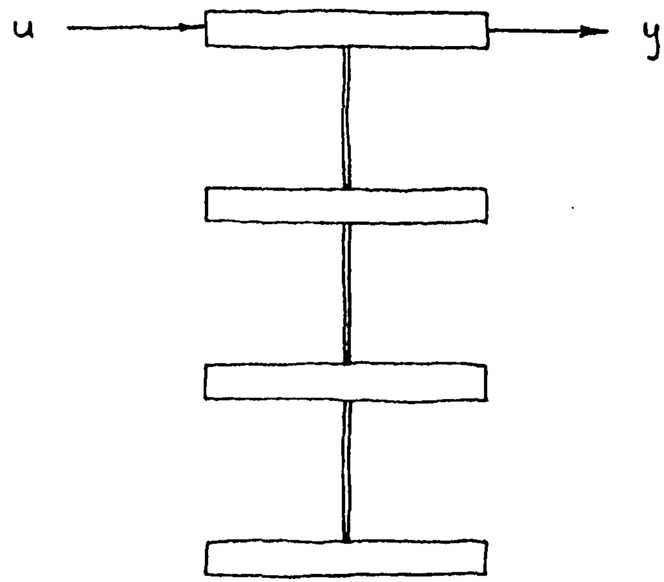
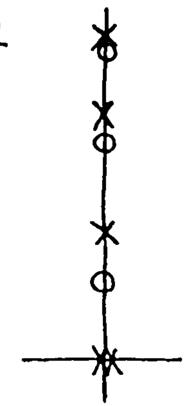
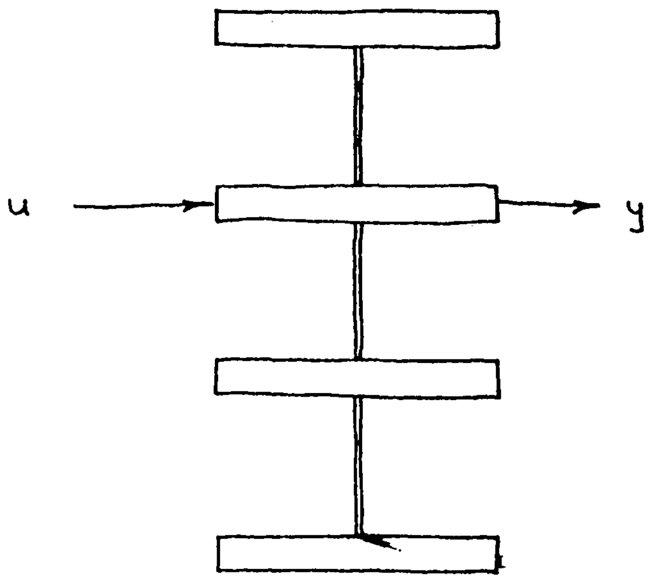


FIG.5 NATURAL MOTION OF PLANT ABOVE



$$\frac{y(s)}{u(s)} = \frac{(s^2+0.198)(s^2+1.555)(s^2+3.247)}{s^2(s^2+2-\sqrt{2})(s^2+2)(s^2+2+\sqrt{2})}$$





$$\frac{y(s)}{u(s)} = \frac{(s^2+0.382)(s^2+1)(s^2+2.618)}{s^2(s^2+2-\sqrt{2})(s^2+2)(s^2+2+\sqrt{2})}$$

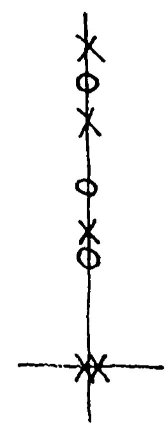
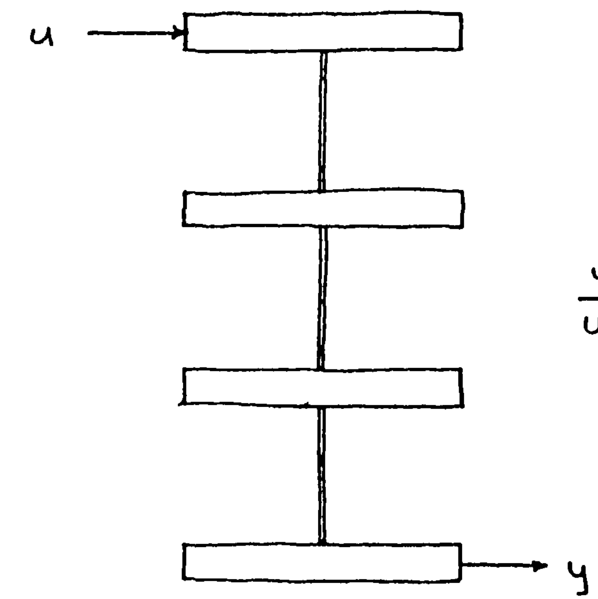


FIG.6a TRANSFER FUNCTIONS OF COLOCATED SENSOR-ACTUATOR PAIRS

FIG.6 TRANSFER FUNCTIONS FOR FOUR-DISK SYSTEM



$$\frac{y(s)}{u(s)} = \frac{1}{s^2(s^2+2-\sqrt{2})(s^2+2)(s^2+2+\sqrt{2})}$$

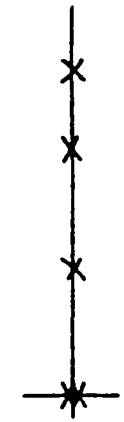
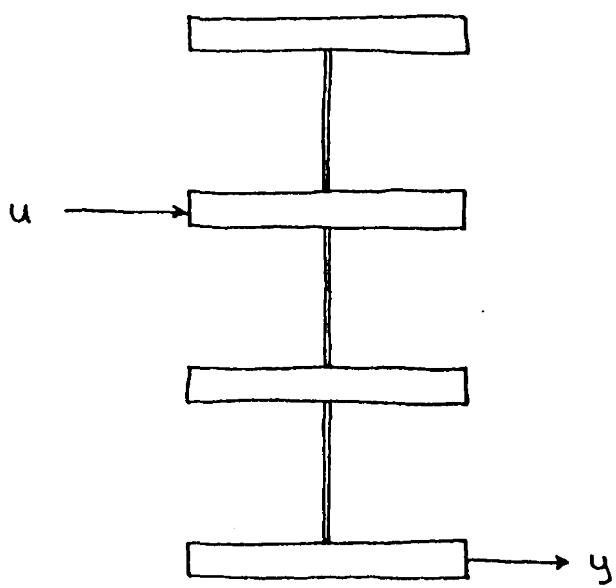


FIG.6b END TO END TRANSFER FUNCTION



$$\frac{y}{u} = \frac{(s^2+1)}{s^2(s^2+2-\sqrt{2})(s^2+2)(s^2+2+\sqrt{2})}$$

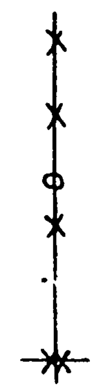


FIG.6c TRANSFER FUNCTION FROM INNER ACTUATOR TO END SENSOR

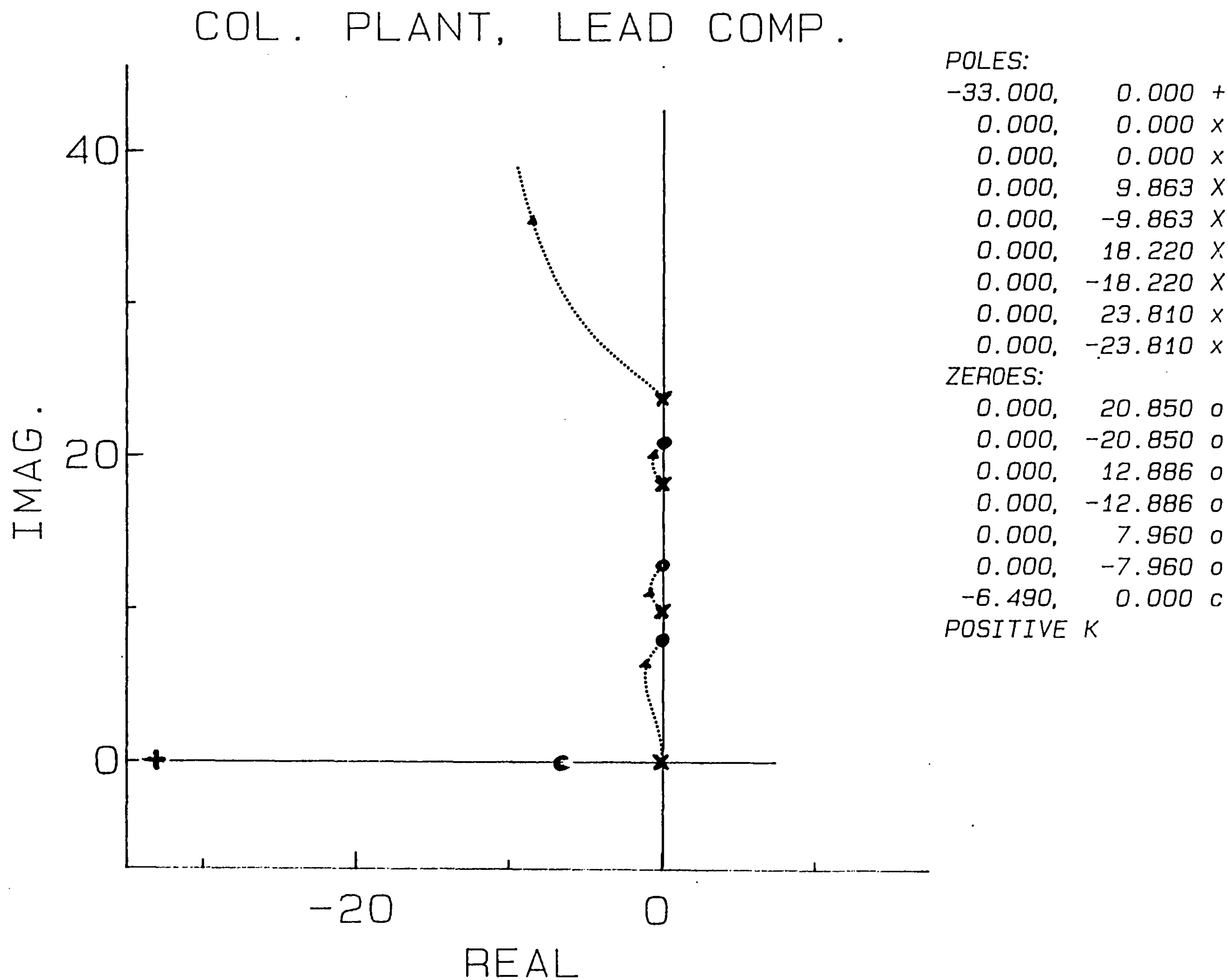


FIG.7 ROOT LOCUS FOR COLOCATED CONTROL USING LEAD COMPENSATION

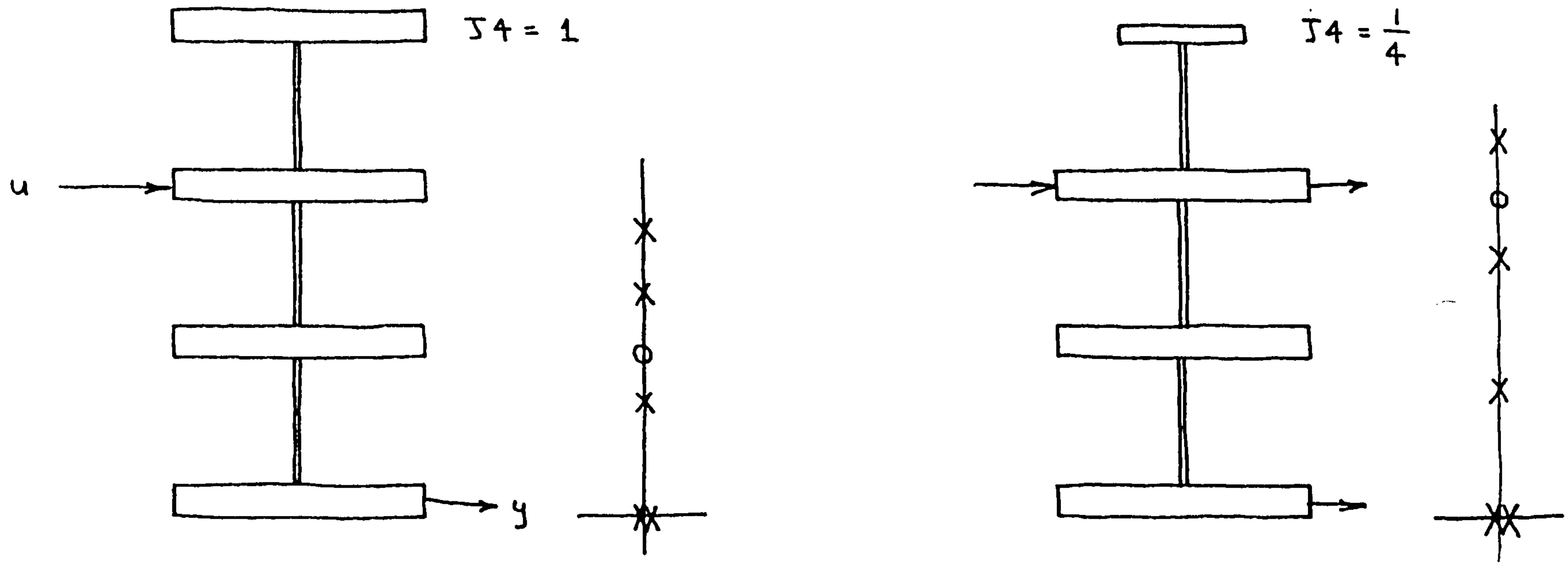


FIG.8 POLE-ZERO FLIP WITH PARAMETER CHANGE IN NONCOLOCATED SYSTEM

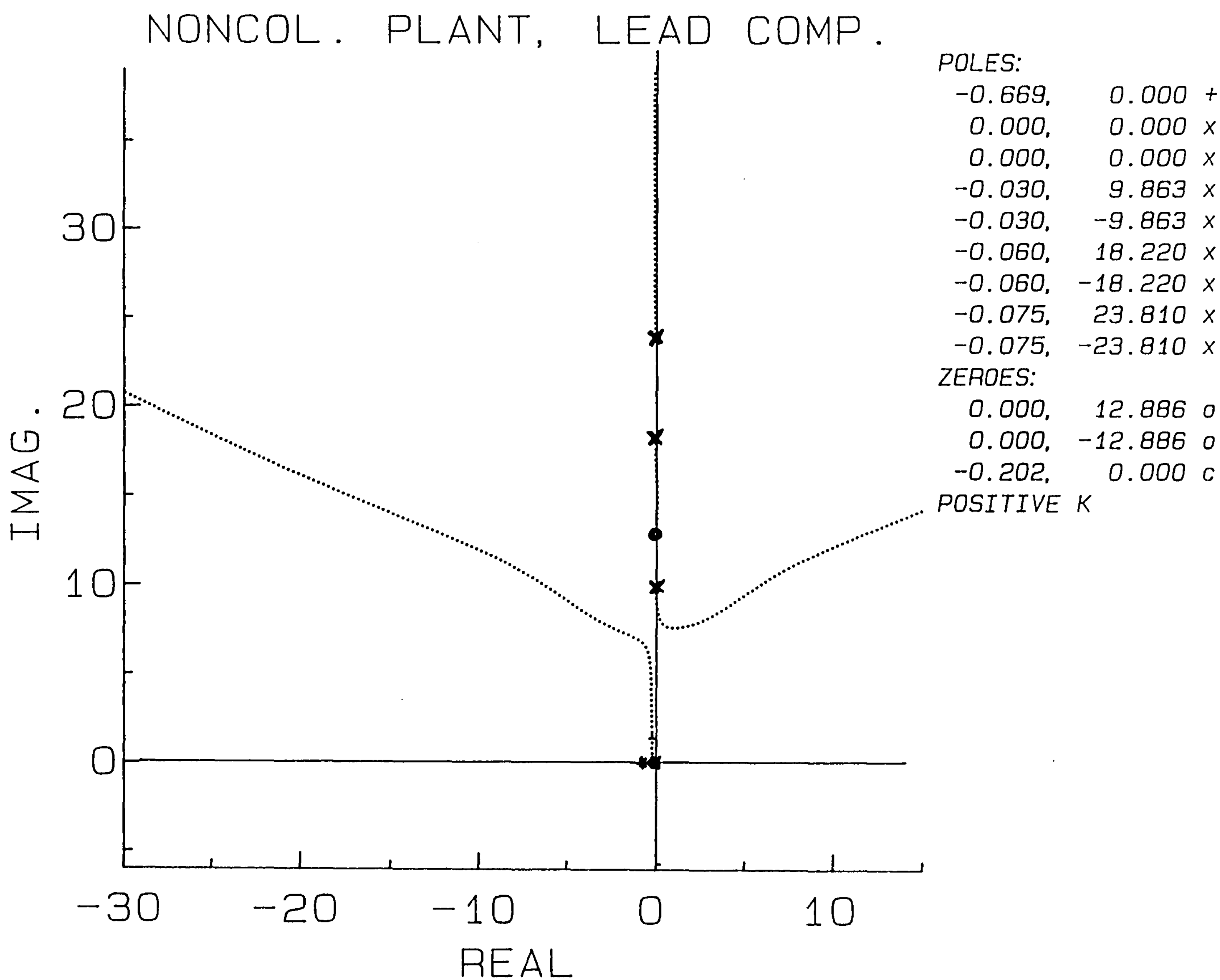


FIG.9 ROOT LOCUS FOR NONCOLOCATED CONTROL USING LEAD COMPENSATION

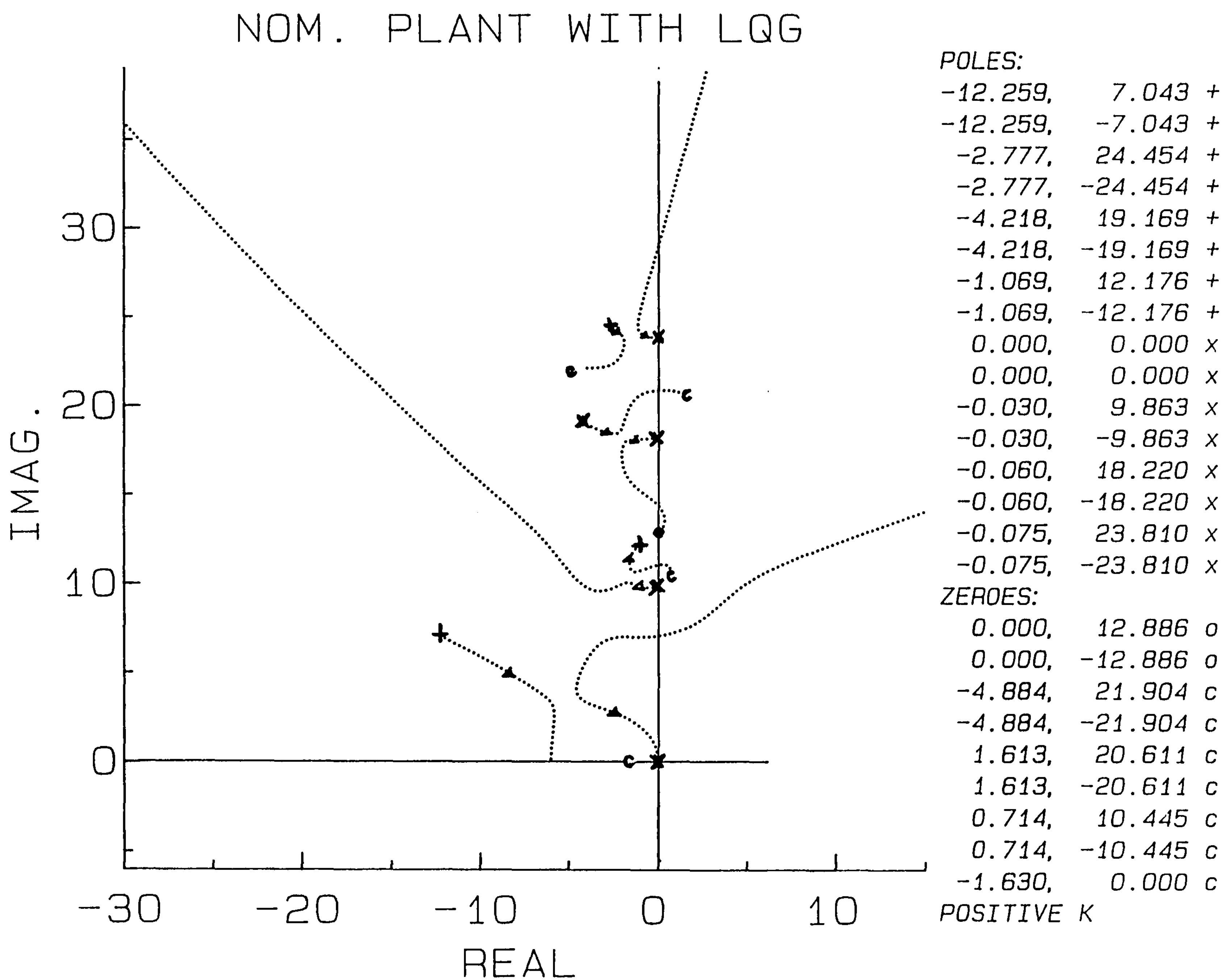


FIG.10 ROOT LOCUS FOR NONCOLOCATED CONTROL USING 8TH ORDER OPTIMAL COMPENSATION: NOMINAL PLANT

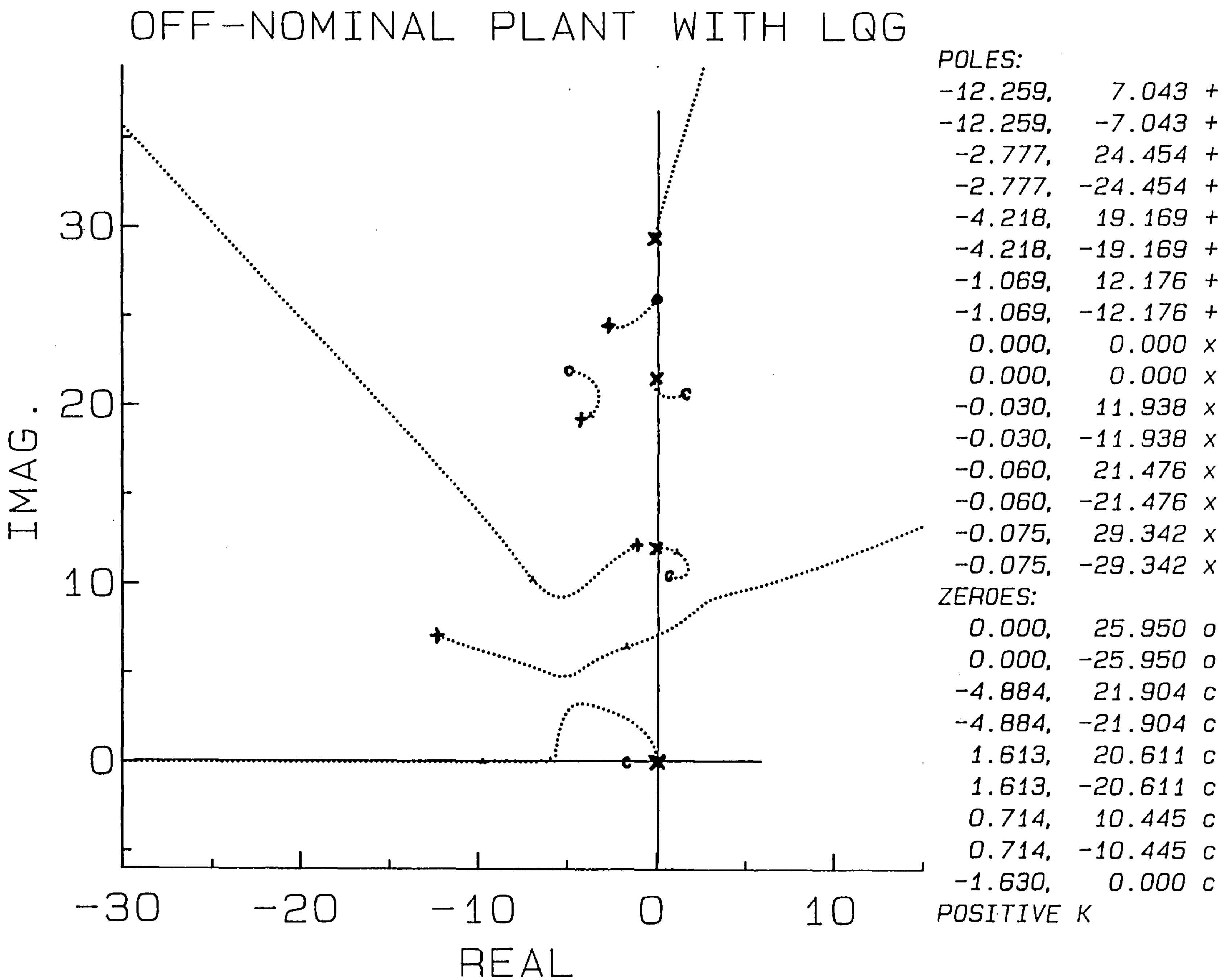


FIG.11 ROOT LOCUS FOR NOMINAL OPTIMAL COMPENSATOR APPLIED TO OFF-NOMINAL PLANT

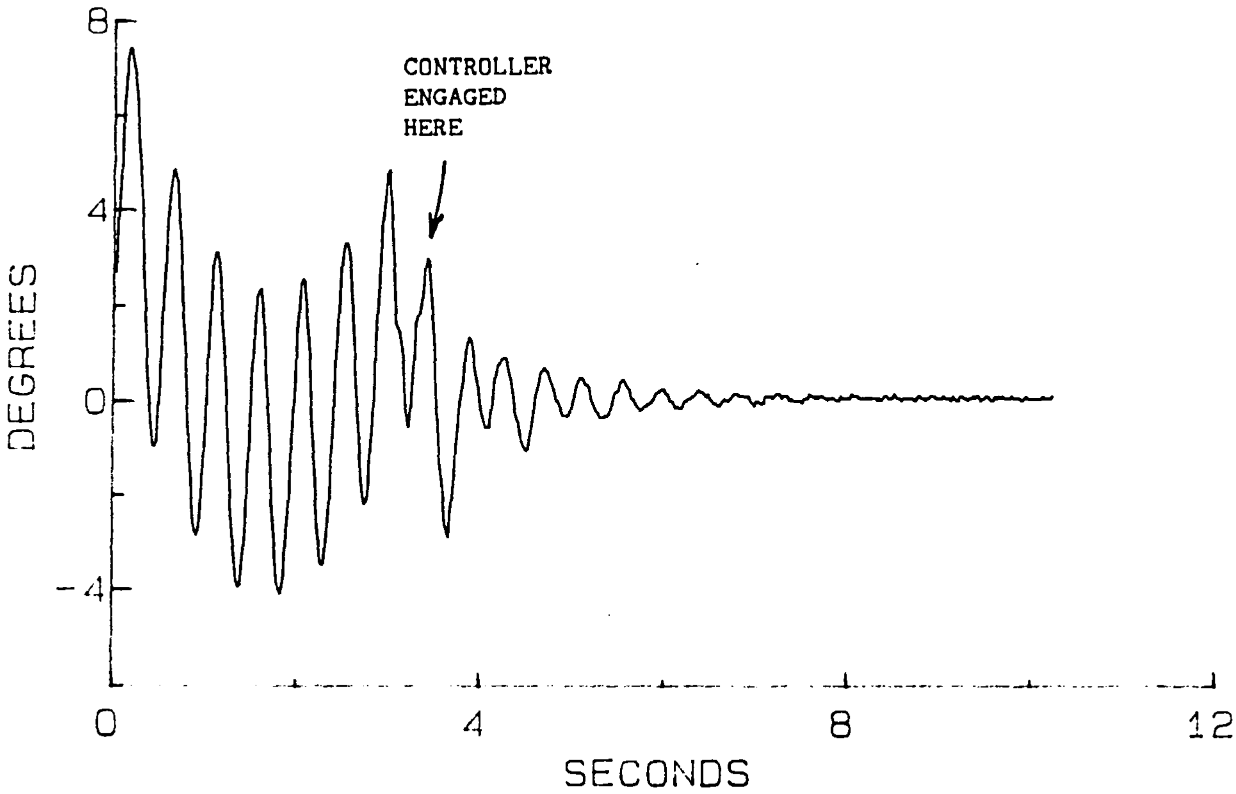


FIG.12a NOMINAL PLANT (RESPONSE TO FIRST MODE INITIAL COND.)

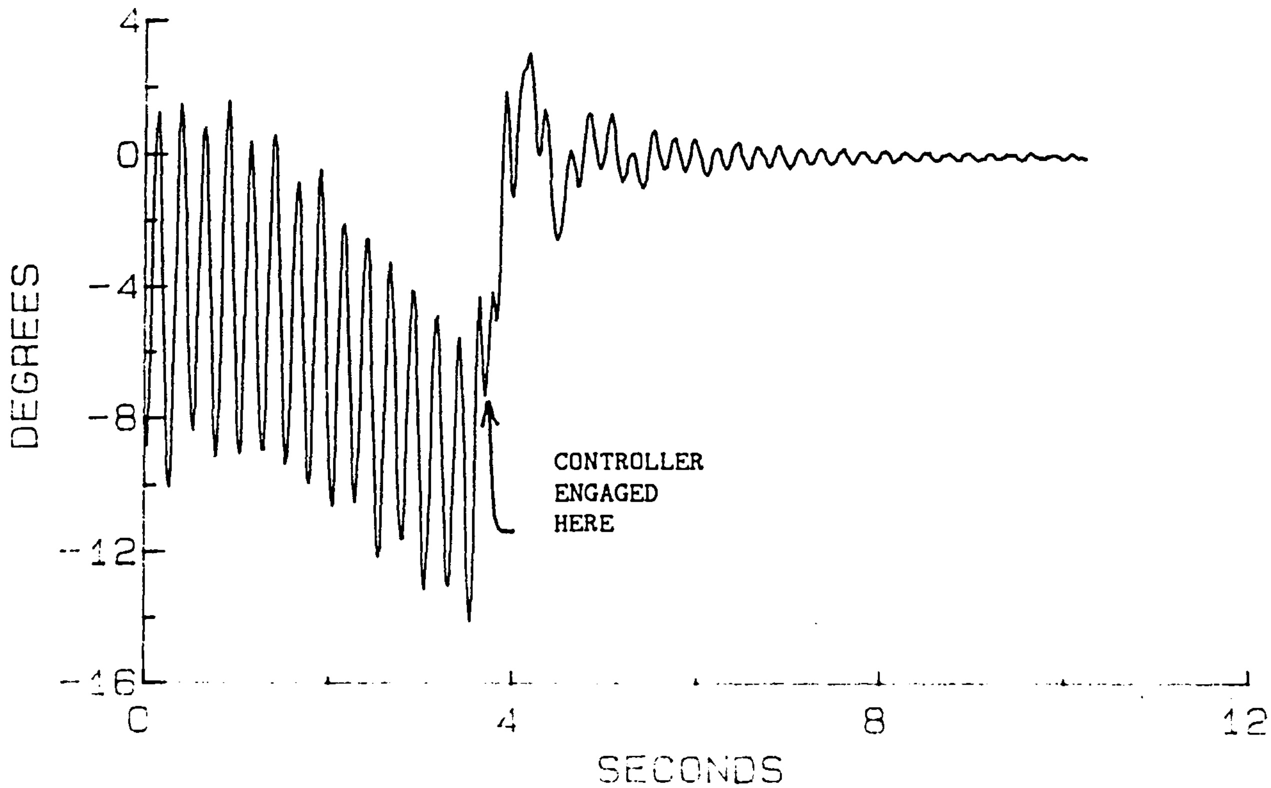


FIG.12b NOMINAL PLANT (RESPONSE TO SECOND MODE INITIAL COND.)

FIG.12 EXPERIMENTAL RESPONSE WITH COLOCATED CONTROL USING LEAD COMPENSATION

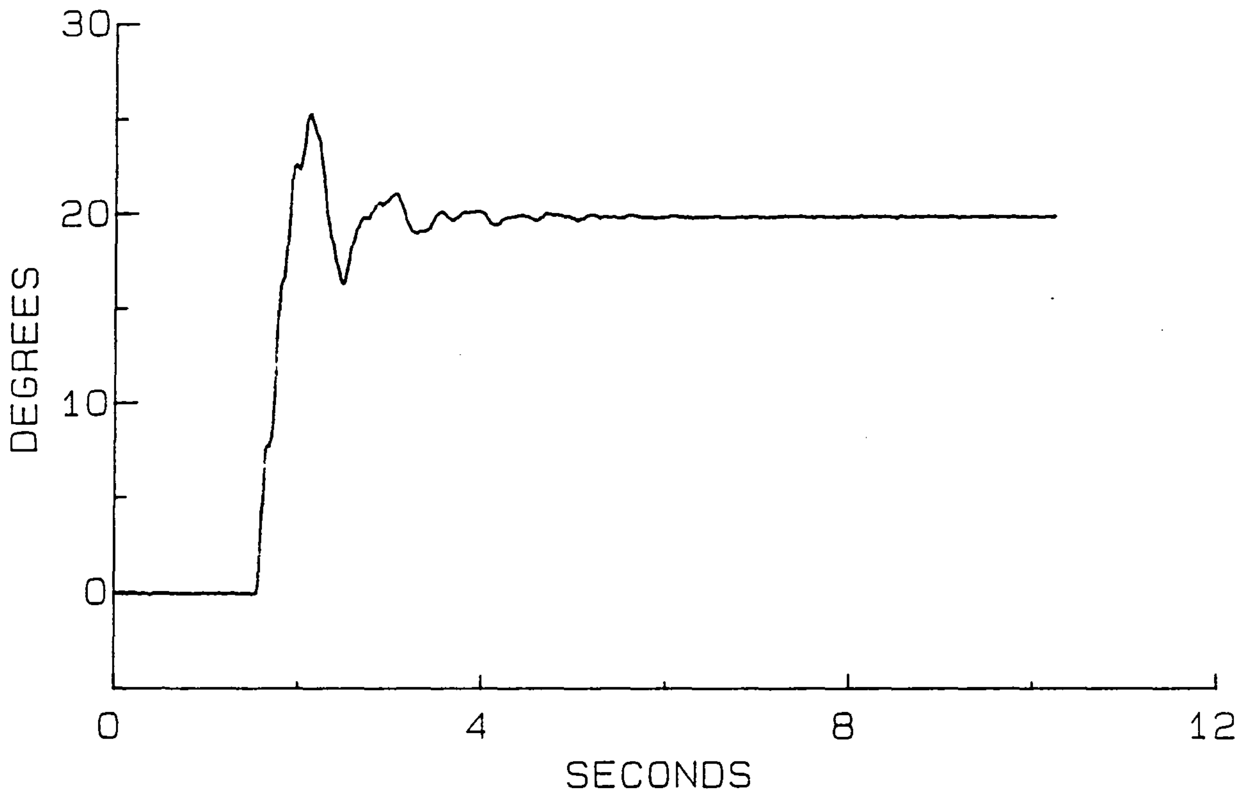


FIG.12c NOMINAL PLANT (RESPONSE TO STEP COMMAND)

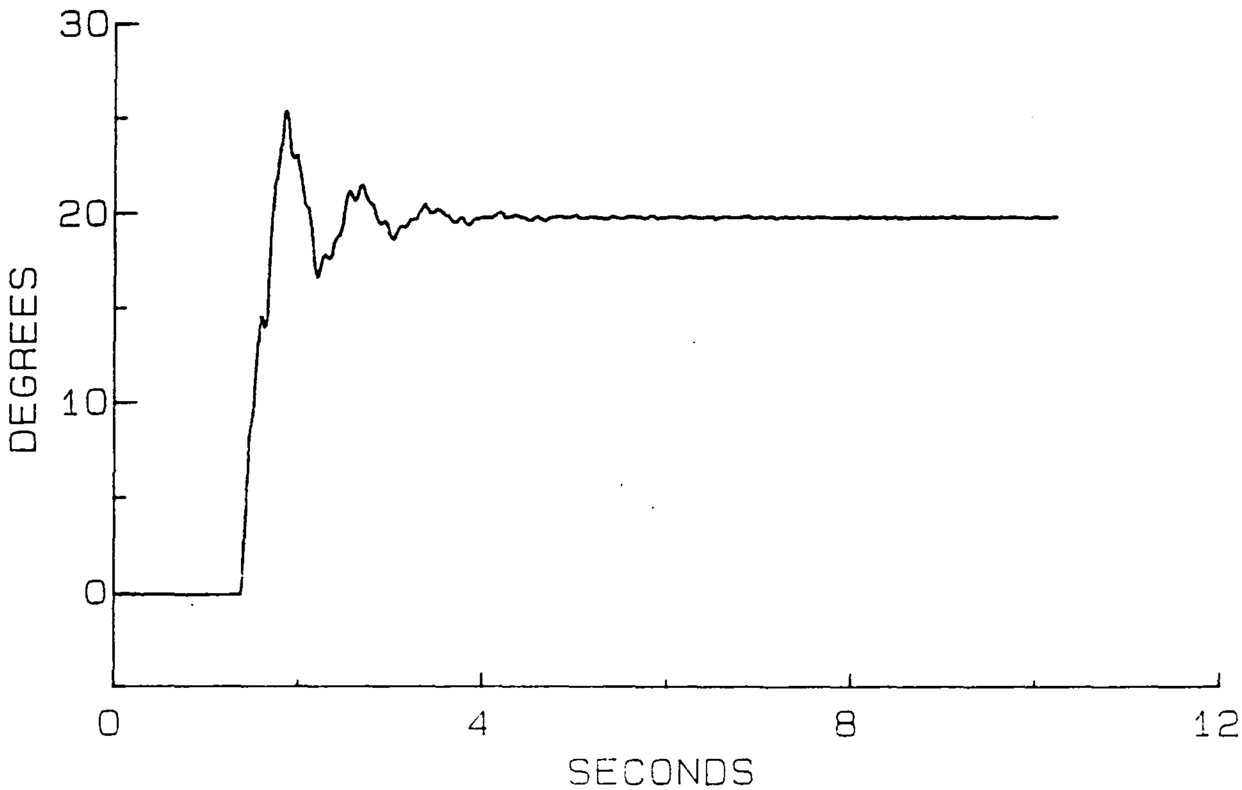


FIG.12d OFF-NOMINAL PLANT (RESPONSE TO STEP COMMAND)

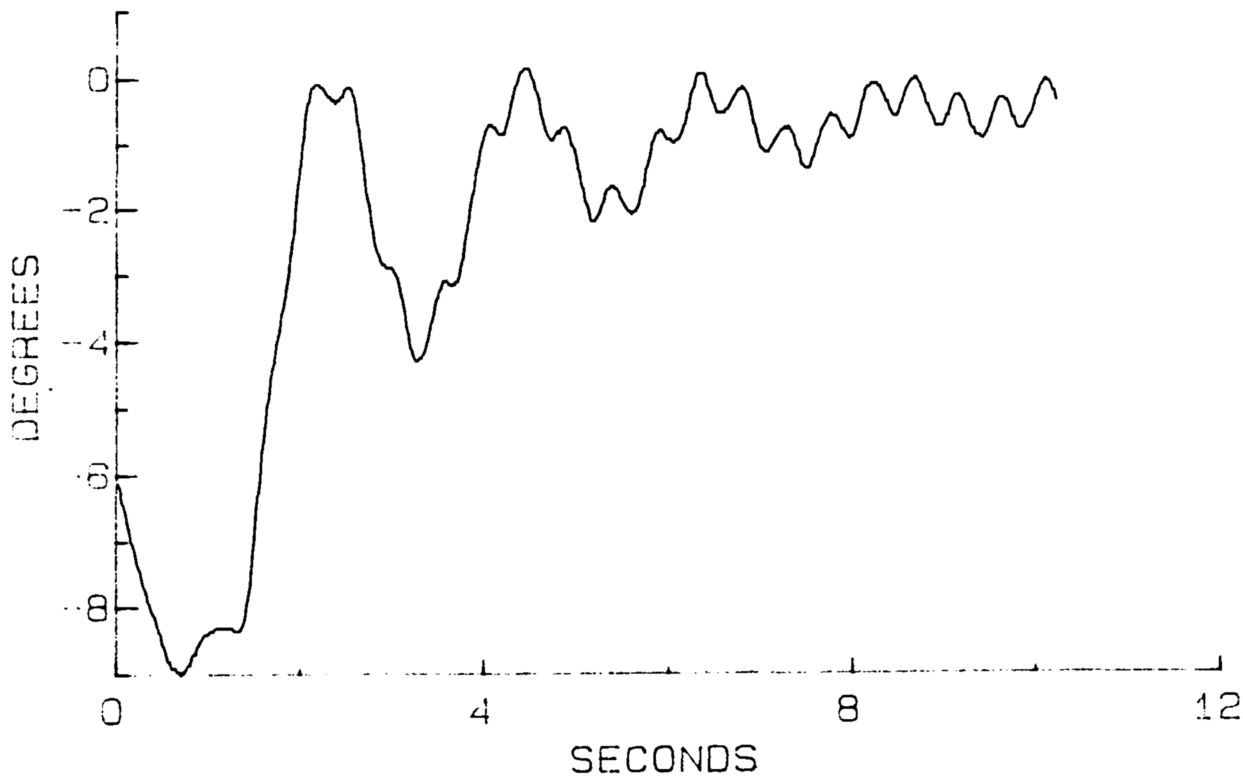


FIG.13a NOMINAL SYSTEM WITH LEAD COMPENSATION

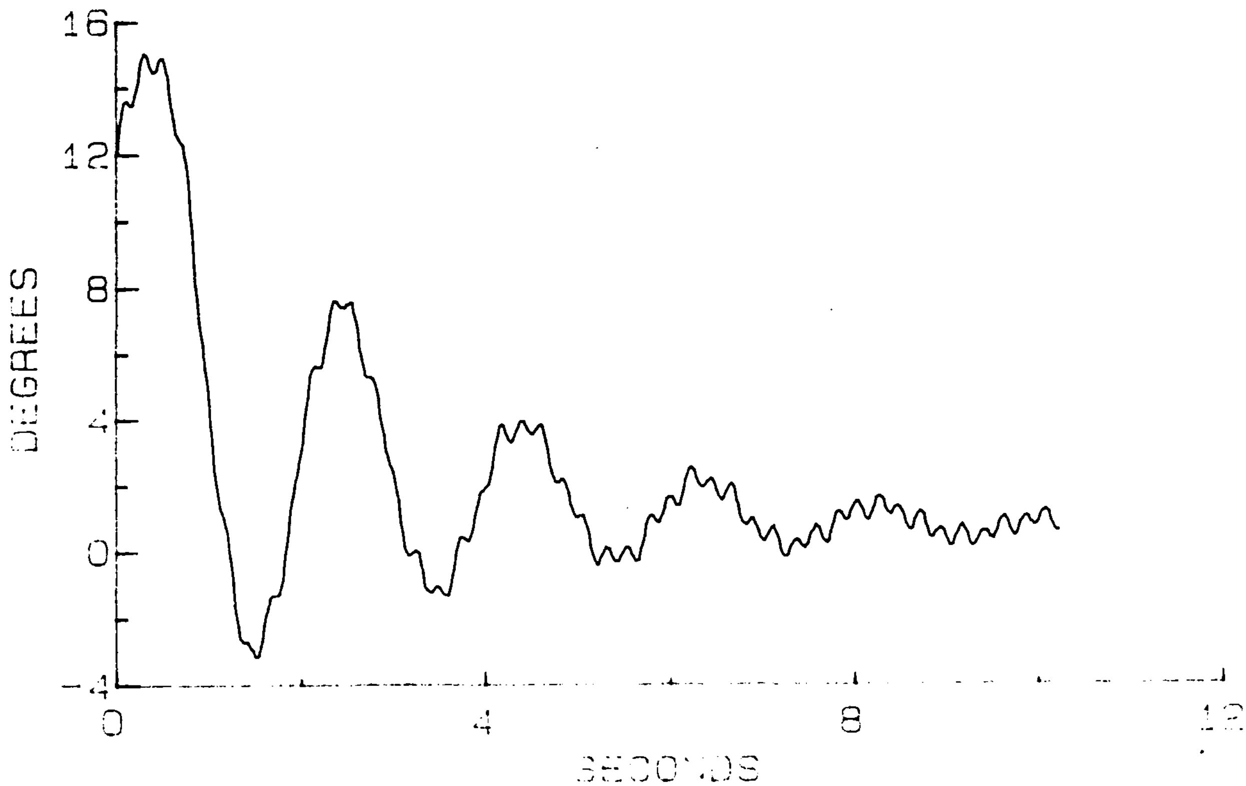


FIG.13b OFF-NOMINAL SYSTEM WITH NOMINAL LEAD

FIG.13 RESPONSE OF NONCOLOCATED SYSTEM, SHOWING SLOW RESPONSE

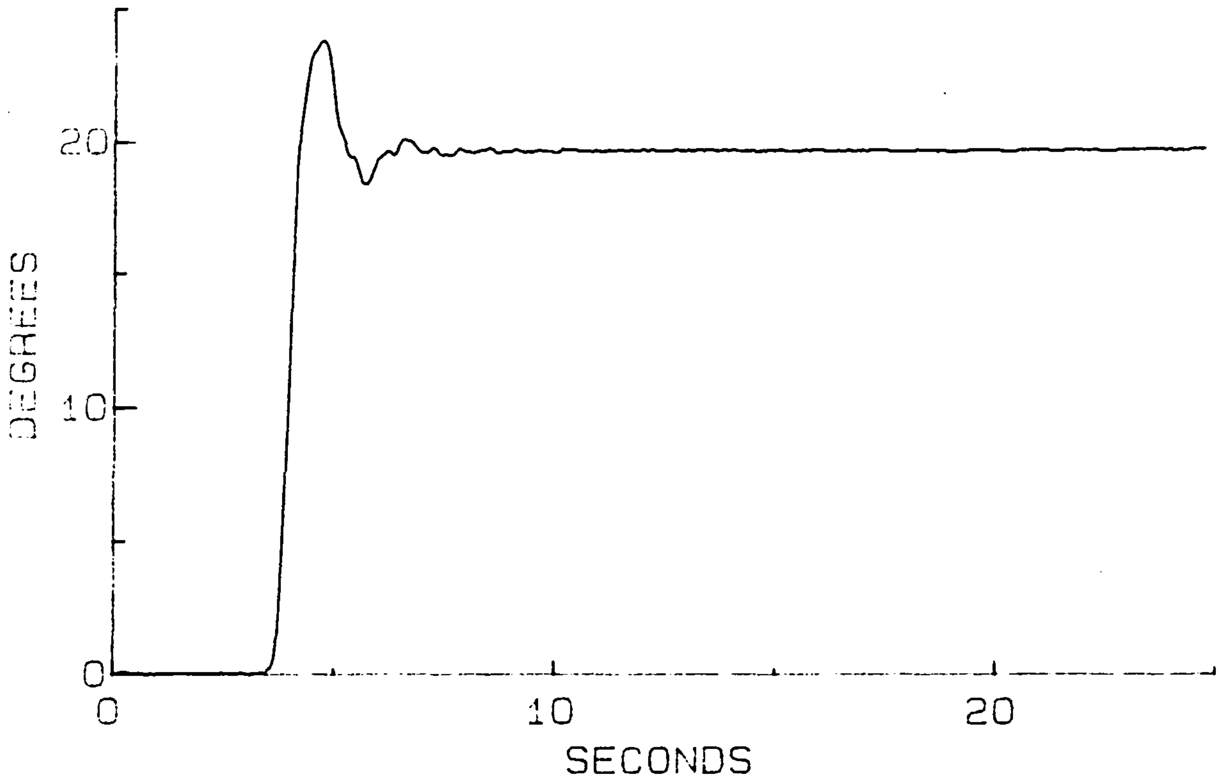


FIG.14a NOMINAL PLANT WITH NOMINAL COMPENSATOR

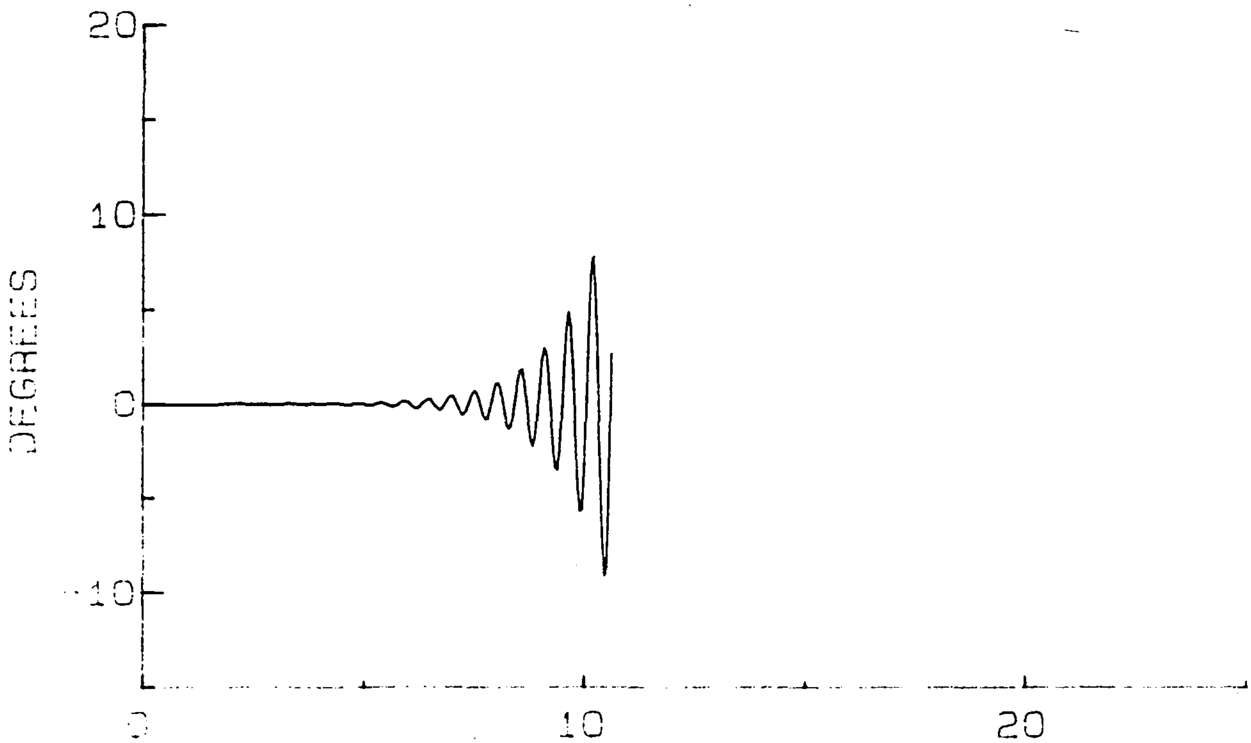


FIG.14b OFF-NOMINAL PLANT WITH NOMINAL COMPENSATOR, SHOWING INSTABILITY

FIG.14 RESPONSE OF NONCOLOCATED SYSTEM WITH 8TH ORDER LQG COMPENSATOR

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